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DETERMINATION OF THE BOUNDARY CONDITIONS AT THE SURFACE OF BODIES SUBJECTED TO ALTERNATING THERMAL EFFECTS

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16. Abstract

A semiempirical method of determining the local thermal fluxes and heat transfer coefficients for turbine blades and similar elements is proposed. It is based on the use of thin film thermocouples, which do not distort the heat transfer conditions and the temperature field at the surface, and on the solution of the inverse nonstationary heat conduction problems.

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Of substantial importance in determination of the temperature fields in /14* the parts of thermal power plants, and the blades of gas turbine engines in particular, is the accuracy of definition of boundary conditions varying both in time and over the surface of a part. There have been many studies devoted to determination of boundary conditions the results of which are characterized by insufficient accuracy. The derivation of third-order boundary conditions is described in the majority of published papers: the temperature of the gas flow around a part and the distribution of the local values of the heat transfer coefficient over the surface of the part. Appropriate corrections must be introduced if the part rotates. For example, rotation of working turbine blading results in substantial variation in α_1 relative to the case of fixed (nozzle) blades [1].

The formulas of G. S. Zhiritskiy and V. L. Lokay, K. Bammert, Pokke and Wilson, and so forth are widely used to find the mean heat transfer coefficients over the blade section outline, α_m. A method is known of determining α_m which is based on analysis of the thermal balance of the gas flow around the blade [2]. The heat exchange conditions depend on the condition of the thermal boundary layer. These conditions may be determined by calculations based on the parameters of the layer in question. This yields approximate criterial relationships for calculation of α_m in blading profiles. Comparative analysis of various formulas for determination of α_m made in [3] reveals a dispersion of values reaching as high as 25%. The dispersion may obviously be even greater in the general case (with the values of α_1 taken as a basis).

Methods have recently been devised for determining the local thermal fluxes and heat transfer coefficients α_1 over the blade outline under full-size conditions by solving the inverse unsteady thermal conductivity problem [4]. The

*Numbers in the margin indicate pagination in the foreign text.

ambient temperature and the temperatures at certain points within the blade are determined empirically. The data obtained are processed by means of simulators, and then values α_1 over the outline yielding the initial temperature values at the thermocouple distribution points are found by means of complicated and laborious operations. Along with the complexity of the adjustment process, for two-dimensional and three-dimensional problems this method requires substantiation that the α_1 distributions obtained are unambiguous. In addition, embedding thermocouples in the body is a complex technological process and results in distortion of the temperature fields.

An advance over these methods is represented by that involving the use of thermometer inserts on the surface of the blades [1]. Even this method, however, is not free of essential deficiencies and restrictions as regards the possible conditions of application.

Hence precise determination of the local boundary conditions on the surface /15 of blades and parts similar to them is a question of great current interest, especially in study under full-scale conditions, under unsteady conditions, and in a number of other important cases. Even greater difficulties arise in the case of disorganized or poorly organized flow of gas around a part, as in thermal fatigue tests of blades or blade models on gas dynamics stands.

To determine the local boundary conditions on the surface of parts we have in the general case employed the surface film thermocouples referred to in [5].

Since the latter are of minimal dimensions (thickness 10-15 μ, width 0.3-0.5 mm), they may be applied in sufficient number to the part being studied without distorting its geometry, the conditions of heat exchange with the gas flux, and the temperature field at the point of insertion of the thermocouple. The temperature measurement error does not exceed 0.5-1%.

Assume that the first-order boundary conditions, $T = T(x, \tau)$, have been found and that the temperature field within the part for any moment in time, $\tau = \tau_n$, has been determined by subsequent calculations or simulation, as has been done, for example, in [5], x being the coordinate a point on the outline of the test piece cross-section.

What is required is determination of the second-order or third-order boundary conditions, i.e., thermal fluxes $q_1 = q(x, \tau_n)$ and heat transfer coefficients $\alpha_1 = \alpha(x, \tau_n)$.

Let us consider the vicinity of point 0 on the outline of the cross-section of a blade or test piece (Figure 1). At time τ_n the temperature of the elementary surface (units) around point 0 is designated as $T_{0,n}$. The temperatures /16 adjacent units are $T_{1,n}$, $T_{2,n}$, $T\ldots$, $T_{i,n}$. The thermal energy transferred from the zeroth unit to the ith over a period of time equalling time interval $\Delta\tau$ is

$$Q_{0i} = \frac{\lambda_{0i} h_{0i} (T_{0,n} - T_{i,n})}{l_{0i}} \Delta\tau,$$

in which λ_{0i} is the thermal conductivity in section 0-i referred to mean temperature $\frac{T_{0,n} + T_{i,n}}{2}$;

l_{0i} is the distance between points;

h_{0i} is the length of the boundary units having 0 and i as centers.

The thermal energy acting to change the heat content in time $\Delta\tau$ is

$$Q_0 = c_0 \gamma_0 (T_{0,n} - T_{0,n-1}) S_0,$$

in which C_0 and γ_0 are the specific heat and the specific gravity of the element around point 0;

S_0 is the area of the element.

The specific thermal flux from the gas toward the surface at point 0 being designated as $q_{1,0}(x, \tau)$, from the thermal balance of the element we obtain

$$q_{1,0}(x, \tau_n) = \frac{c_0 \gamma_0 S_0}{h_0 \Delta\tau} (T_{0,n} - T_{0,n-1}) + \sum_{i=1}^{k} \frac{\lambda_{0i} h_{0i}}{h_0 l_{0i}} (T_{0,n} - T_{i,n}). \quad (1)$$

Here h_0 is the length of a section on the outline of the cross-section;

k is the number of adjacent elementary units (k = 1, 2, 3, ...).

If the temperature field has been determined and the temperature at point 0 is known, the specific thermal flux from the gas toward the blade at point 0 is determined from formula (1). The breakdown of the cross-section into elementary areas (units), and consequently their parameters, remain the same as in determination of the temperature field [5]. Hence a small addition to the

thermal field calculation program permits determination of the local thermal fluxes at points on the outline.

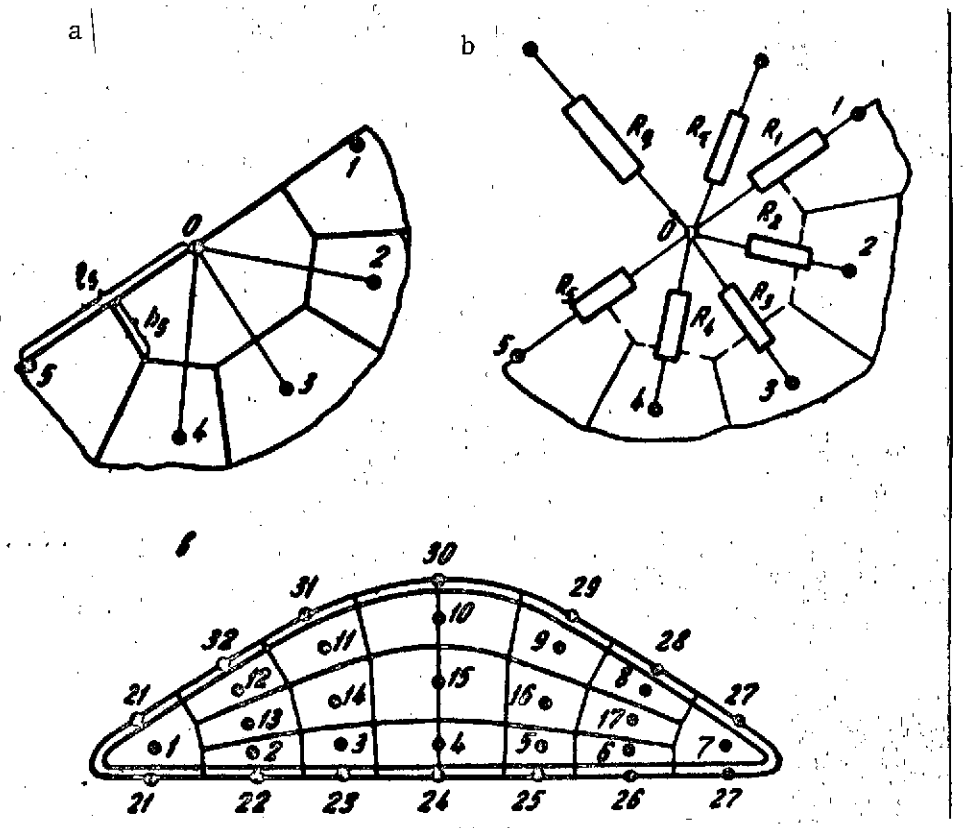

Figure 1. Boundary Section of Cross-Section Outline of a Body (a) with its Electric Model (b) and Breakdown of the Profile Test Piece Cross-Section into Units (c).

The local values of heat transfer coefficients α_{10} at point 0 at known ambient temperature T_a are determined from the formula

$$\alpha_{1.0} = \frac{q_0}{T_c - T_{0,n}}. \qquad (2)$$

Determination of Local Thermal Fluxes q_1 and Local Heat Transfer Coefficients α_1 with Simulators

We write expression (1) in the form

$$q_{1.0}h_0 = \frac{c_0\gamma_0 S_0}{\Delta\tau}(T_{0,n} - T_{0,n-1}) + \sum_{\alpha=1}^{k}\frac{\lambda_{0\alpha}}{l_{0\alpha}}h_{0\alpha}(T_{0,n} - T_{l,n}). \qquad (3)$$

The Kirchhoff law equation for the electric model (Figure 1) of region 0 is written as

$$\frac{V_q - V_0}{R_q} = \frac{V_0 - V_\tau}{R_\tau} + \sum_{i=1}^{k} \frac{V_0 - V_i}{R_i}. \tag{4}$$

The analogy conditions of equations (3) and (4) yield the following expressions for the parameters of the model:

$$R_\tau = \frac{\Delta\tau}{c_0 \gamma_0 S_0} R_N;$$

$$R_i = \frac{l_{0i}}{\lambda_{0i} h_{0i}} R_N. \tag{5}$$

in which R_N is a normalizing factor.

In this case

$$q_{1.0} = \frac{V_q - V_0}{R_q h_0} (kR_N), \tag{6}$$

k being the scale of transition from temperature to voltage values $\left[\frac{\text{degrees}}{\text{v}}\right]$. If we set $R_q = \frac{1}{h_0} R_N$, then it follows from formula (6) that

$$q_{1.0} = (V_q - V_0) k. \tag{7}$$

Hence in order to determine the local thermal flux value at a point on surface 0 it is necessary, for time $\tau = \tau_n$,

1) to supply to resistance R_i potentials V_t corresponding to the temperature of the adjacent units, and to resistance R_τ potential V_τ corresponding to the temperature of point 0 at the preceding moment in time;

2) to vary potential V_q on resistance R_q in order to establish at point 0 potential V_0 corresponding to the temperature at point 0 at the nth moment in time.

According to formula (7), the potential difference on resistance R_q equals thermal flux $q_{1.0}$ on the scale selected. If the potential on resistance R_q exceeds the capacity of the integrator, it is then necessary to establish on resistance R_q any potential difference (positive if the body is heated and

negative if it is cooled) and to bring about the corresponding potential at point 0. In this case the value of $q_{1.0}$ is calculated from formula (6).

The values of α_1 may be determined analogously with the model with ambient temperature T_a known. It is necessary for this purpose to supply to resistance R_q the potential corresponding to the ambient temperature and to vary R_q so as to establish at point 0 the potential corresponding to the temperature at this point $T_{0,n}$. The value of the heat transfer coefficient at point 0 is determined from the formula

$$\alpha_{1.0} = \frac{1}{R_q h_0} R_N. \tag{8}$$

The relationships cited are valid for unsteady or steady thermal conditions. In the case of steady conditions it is more convenient to employ models made of the electrically conducting paper included in the set with the EGDA-9/60 integrator.

Let us consider as an example the case of determination of the second-order and third-order local boundary conditions in thermal fatigue testing of profile test pieces, models of gas turbine engine blades. The test pieces are subjected to heating in kerosene combustion products, and then to cooling in a flow of air. In this particular case it is virtually impossible to evaluate the conditions of heat exchange between the test piece and the flow. We measured the average surface temperatures over the height of a test piece cross-section by means of film thermocouples and then calculated the temperature fields in the cross-section at various times of the heating-cooling cycle [5].

The thermal fluxes were determined at outline points 21-33 of the test piece cross-section (Figure 1c) for all time intervals, on the basis of the method discussed in the foregoing and with use made both of the model and digital computer calculations. The variation in the thermal flux value at points 24, 29, and 27 with the time of the cycle is illustrated in Figure 2. The /18
temperature variation curves at these points obtained by film thermometry are also presented in this illustration. Determination of the thermal fluxes along the broken-line sections was complicated by the circumstance that at these times the test piece was transferred from the kerosene burner flame to the cooling air flow.

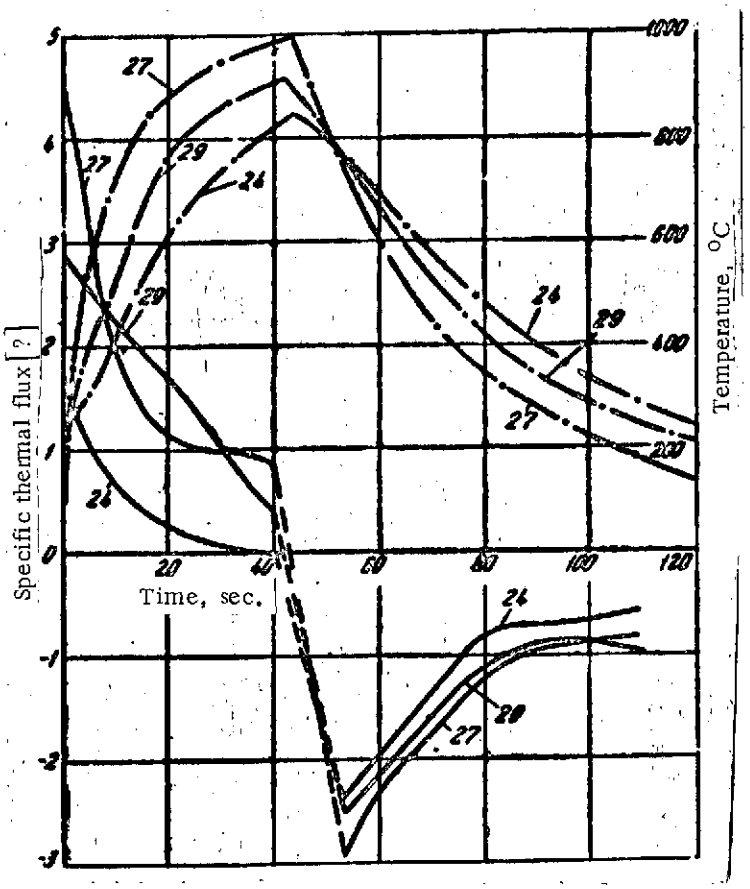

Figure 2. Variation in Local Thermal Fluxes q_1 in Heating-Cooling Cycle and in Surface Temperature at Points 24, 27, and 29 of Cross-Section of Profile Test Piece.

Figure 3 illustrates the distribution of local thermal fluxes over the outline of a test piece cross-section at certain moments in time on heating (a) and cooling (b).

The local values of the coefficient of heat transfer α_1 from the medium to the sample is determined on the basis of the measured ambient temperature and cooling air temperature, t_a = 1500°C and $t_{c.air}$ = 20°C. The distribution of α_1 over the outline of the cross-section, which is illustrated in Figure 4, remains in the first approximation uniform throughout the heating (2) and cooling (1) half-cycles. The data obtained on the basis of α_1 reveal considerable differences in the average value of heat transfer coefficient α_a (an approximately twofold difference) for the heating and cooling half-cycles.

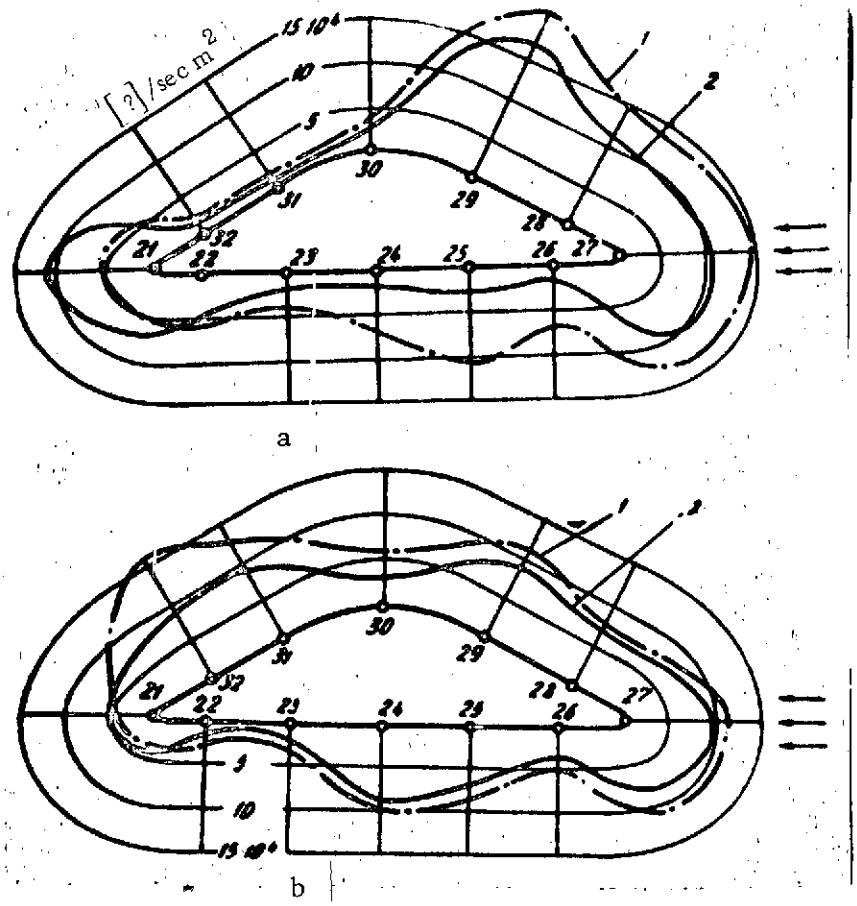

Figure 3. Distribution of Local Thermal Fluxes q_{ln} over Surface of Profile Test Piece During Heating (a) and Cooling (b) Half-Cycles. /19

In order to evaluate the certainty of this difference analysis was made of the case of heating-cooling (Figure 5) of an infinitely long body heated uniformly over the cross-section. The thermal conductivity equation is in this case written in the form

$$\frac{d\bar{U}}{dFo} = K\,Bi\,(\bar{U}_c - \bar{U}), \tag{9}$$

where

$$\bar{U}_c = \frac{t_c - T_{min}}{T_{max} - T_{min}};$$

$$\bar{U} = \frac{t - T_{min}}{T_{max} - T_{min}};$$

Fo, Bi are the Fourier and Biot numbers;

$K = \frac{P}{S}$ is the form factor;

S and P are the cross-sectional area and perimeter;

Z is the typical linear dimension (diameter);

T_{max} and T_{min} are the maximum and minimum temperatures of the cycle, which are established automatically;

t_a and $t_{c.air}$ are the temperatures of the hot flow and the cooling air;

t is the temperature of the body.

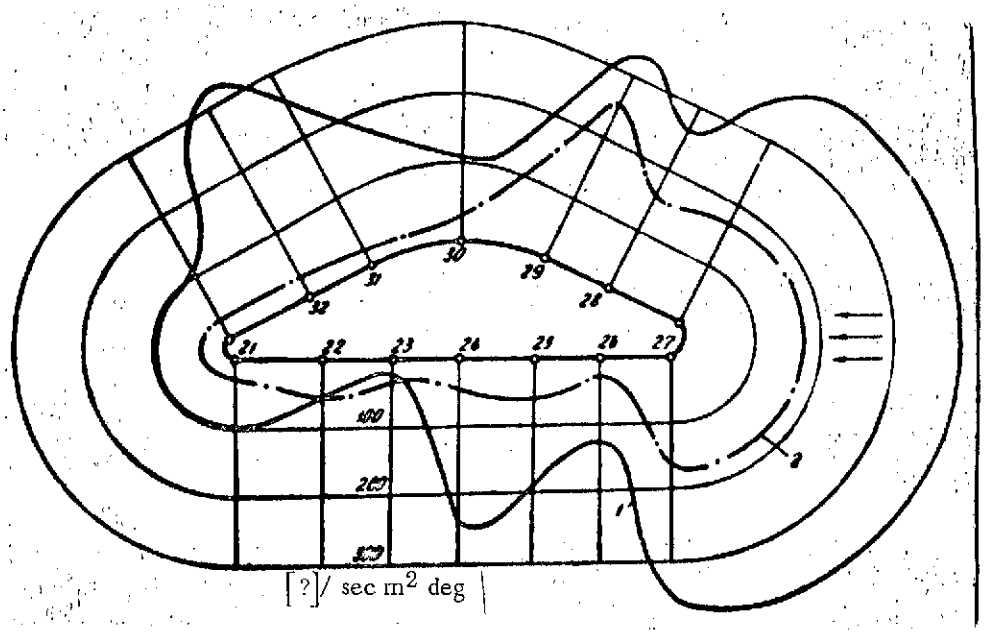

Figure 4. Distribution of Local Heat Transfer Coefficients Over Cross-Section of Profile Test Piece During Heating (2) and Cooling (1) Half-Cycles.

The solution of equation (9) is of the form

$$\bar{U} = (\bar{U}_0 - \bar{U}_c)\, e^{-K\,Bi\,Fo} + \bar{U}_c. \tag{10}$$

The value of Fo_{heat}, which corresponds to the time at which the body reaches T_{max}, i.e., when $\bar{U} = 1$, may be determined from formula (10):

$$Fo_{heat} = \frac{1}{K\,Bi_{heat}} \ln \frac{\bar{\Delta}_{heat}}{1 + \bar{\Delta}_{heat}},$$

where

$$\bar{\Delta}_{heat} = \frac{t_c - T_{max}}{T_{max} - T_{min}}. \tag{8'}$$

Similarly for the cooling half-cycle:

$$Fo_{cool} = K \frac{1}{Bi_{cool}} \ln \frac{\bar{\Delta}_{cool}}{1 + \bar{\Delta}_{cool}},$$

where /21

$$\bar{\Delta}_{cool} = \frac{T_{min} - t_{cool}}{T_{max} - T_{min}}.$$

On the basis of the condition of equality of duration of the half-cycles the value of Fo_{heat} is set to equal that of Fo_{cool}, and the following relation is obtained:

$$\frac{Bi_{heat}}{Bi_{cool}} = \frac{\ln \frac{\bar{\Delta}_{heat}}{1 + \bar{\Delta}_{heat}}}{\ln \frac{\bar{\Delta}_{cool}}{\bar{\Delta}_{cool} + 1}}. \qquad (11)$$

Formula (11) permits determination of the relationship between the average heat transfer coefficients during heating and cooling, which equals the Bi ratio.

The parameters characterizing the cycle in question for a profile sample (Figure 5) being specified as follows: $T_{max} = 900°C$, $T_{min} = 200°C$, $t_a = 1500°C$, and $t_{c.air} = 20°C$, it is found from formula (11) that $\alpha_{heat} \simeq 0.5\ \alpha_{cool}$. The result obtained agrees with the data derived from empirical determination of α_1 for a profile test piece (Figure 4) and provides confirmation that the analyses were correctly performed.

A matter of definite interest is that of the possibility of calculating thermal fluxes in parts of finite geometry with a formula based on measurement of the surface temperature of a semi-infinite body. Thermal flux sensors of this type are described in [6]; they are employed chiefly in shock tube measurements. We have performed the corresponding calculations for points 24, 27, and 29 on the outline of a profile test piece. The temperature at these points was approximated in time by the exponent

$$t(\tau) = t_0 + A[1 + \exp(-b\tau)]. \qquad (12)$$

The design equation is then of the form

$$q = 2A \sqrt{\frac{\overline{\lambda c \gamma}}{\pi}} \sqrt{b} \exp(-b\tau) \int_0^{\sqrt{b\tau}} \exp Z^2\, dZ. \qquad (13)$$

λ, c, γ are the specific thermal conductivity, specific heat capacity, and specific gravity of the material of the part.

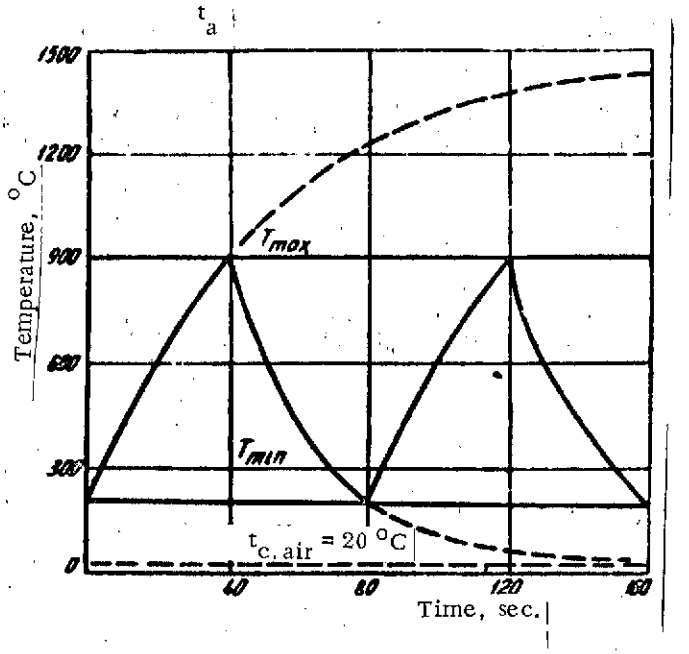

Figure 5. Determination of Bi_{heat}/Bi_{cool} Ratio for a Body Subjected to Heating and Cooling Cycles.

The thermal fluxes calculated from formula (13) exhibited a manifold difference from the actual values. This indicates the impossibility of employing this method in the case in question, in which the time of heating (or cooling) greatly exceeds "characteristic" time $\tau = \frac{Z^3}{4a}$, where a is the coefficient of thermal conductivity.

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